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The choice of the transmission structure of a wheeled machine for the mechanization of thinning

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Abstract. The article reflects the relevance of the mechanization of work and individual operations in the course of planned thinning in forests of various types. The specific operating conditions of machines used for this type of forestry work are determined. The principles of construction of transmissions of machines for various purposes are considered, a method for determining external parameters and constructing a kinematic diagram of an inter-axle controlled power distribution mechanism in its composition is proposed. The results of kinematic and force analysis for the considered example of the mechanism are presented. Prospects for using the mechanisms of this family are formulated.

1 Introduction

An analysis of the experience of the forest industry and modern scientific publications on sustainable forest management [1-7] show the high importance of the rational organization of activities related to the maintenance and restoration of the forest fund. One of the central places in terms of forestry work is the organization and provision of thinnings [8, 9]. It is known that planned thinning helps reduce the likelihood of the spread of forest fires and simplifies the work on localizing and extinguishing them, and also helps to increase the productivity of forest crops.

Technological features of thinning require both the use of small-scale mechanization [10] and the use of full-size wheeled and tracked chassis equipped with specialized technological equipment (hydraulic manipulator, harvester head, grab, mulcher, etc.) and capable of transporting harvested timber [8, 9]. The mass of such machines is 10-13 tons, and the chassis has a rigid or articulated design. For example, the chassis of the harvester «Highlander» has three drive axles, two of which are arranged in such a way as to allow the installation of removable tracks if necessary.

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The issues of interaction between the mover of forest machines and soil have been considered by many researchers [11]. At the same time, the problem of rational transmission design for such machines also needs to be studied. So, about 40% of the working time of the chassis, occupied in the transportation of timber, falls on idling [12]. In this case, the normal reactions under the front drive axle and rear axles can be distributed almost equally, and when the chassis is fully loaded, the rear axles are often overloaded. Redistribution of the load occurs during skidding of wood and towing, as well as when overcoming slopes. Thus, during 40-50% of the working time, the front and rear axles can realize approximately the same traction force in terms of adhesion to the ground, and in the remaining time it is more profitable to provide an asymmetric distribution of the moment.

In modern mechanical transmissions of traction and transport machines, the problem of controlling the values of torques transmitted to the drive axles is not solved. In practice, hydromechanical transmissions are used, and electromechanical power transmissions also have certain prospects. But mechanical transmission has the advantages of simplicity, reliability, high energy consumption, low cost of manufacture and operation. In addition, technologies for the manufacture of mechanical transmissions for tractors and cars have been well tested in Russia. The development of a node that provides control of the power flows supplied to the drive axles would make mechanical transmissions more attractive to the consumer. This node is called a controlled power distribution mechanism and is part of the transfer case or completely replaces it functionally.

The purpose of the study is to improve the operational properties (cross-country ability, mobility, traction and dynamic capabilities) of forest wheeled vehicles using a controlled inter-axle power distribution mechanism.

The objects of research are the transmissions of wheeled forest machines.

Research methods – methods of theoretical mechanics, the theory of transport caterpillar and wheeled vehicles, the theory of the tractor.

2 Problem statement

Working with literary sources and analyzing the experience of designing and operating the chassis of transport and traction machines [1-10, 12] allow us to formulate the following tasks.

1. To analyze the specifics of the use of wheeled vehicles in forestry during the planned thinning.
2. Propose an approach to building a transmission structure that allows for control over the redistribution of torque between the drive axles.
3. Suggest a method for choosing the main parameters and constructing a kinematic diagram of a controlled inter-axle power distribution mechanism in such a transmission.
4. Adapt the methodology of kinematic and force analysis of planetary gears to the case of the family of mechanisms under consideration and determine the prospects for their use.

3 Results and discussion

Based on the analysis of the operating conditions and the purpose of the chassis, the following transmission structure is proposed: a connecting mechanism; transforming mechanism; controlled inter-axle power distribution mechanism (PDM); cardan gears; main gears, cross-axle differentials, axle shafts and wheel gears as part of drive axles.

It is economically expedient to perform the transforming mechanism on the basis of a shaft gearbox. Controlled inter-axal PDM is performed on the basis of a planetary gearbox.

The traditional approach to the synthesis of kinematic schemes of planetary mechanisms [13-15] implies the principle of providing given gear ratios between the driving and driven links. But the inter-axle PDM has two driven links. In addition, during normal operation of the planetary gearbox, one degree of freedom is realized, and during the operation of the PDM, two, as in the case of an inter-wheel power distribution mechanism [16-19].

An uncontrolled PDM should be taken as a basis, which allows one of the desired operating modes to be implemented (for example, an asymmetric differential with a kinematic parameter (–2), torque distribution 1: 2) and modified based on the principle of enabling the second operating mode (for example, torque distribution at the rate of 1:1). To do this to have to use a connecting link, an additional planetary gear set and at least two controls. The mechanism will have three degrees of freedom; when using controls, the number of degrees of freedom is reduced to two. Additional opportunities open up when using braking of the connecting link - the number of degrees of freedom of the MRM is reduced to one, that is equivalent to disabling one of the bridges.

An additional row can be connected to any of the output branches of the PDM, but in order to provide the option to turn off the front axle, the option with the modification of the "front" branch is promising.

The kinematic diagram of the mechanism is shown in Figure 1.

The developed mechanism has three degrees of freedom, consists of five main links and in the basic version uses two control elements (brake and blocking clutch).

The asymmetric differential, which is the basis of the PDM, is characterized by the kinematic parameter $k_1 = (–2)$ and gear ratios (Figure 1):

$$u_{0X} = \omega_0/\omega_X = -M_X/M_0 = 2/3 \tag{1}$$

$$u_{0A} = \omega_0/\omega_A = -M_A/M_0 = 1/3 \, . \tag{2}$$

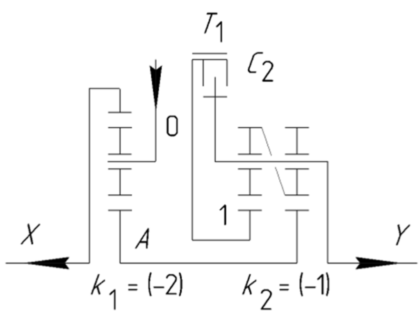

Fig. 1. Kinematic diagram of a controlled inter-axle PDM: 0, X and Y – leading and driven links; A and 1 – connecting and brake links; T1 and C2 – controls.

Thus, the gear ratio of the additional row built into the branch associated with the front drive axle will be:

$$u_{AY} = \omega_A/\omega_Y = -M_Y/M_A = u_{0X}/u_{0A} = 2 \, . \tag{3}$$

Planetary gear sets that allow such a gear ratio are characterized by kinematic parameters equal to (–1) and (+2). On Figure 1 shows the option of including a series with k2 = (–1) in the PDM scheme.

The following main modes of operation are implemented:

- neutral (all controls are disabled, power transmission to drive axles is not possible);
- asymmetric distribution of moments (blocking clutch C_2 is on);
- symmetrical torque distribution (brake T_1 is on).

Additionally, by introducing the appropriate elements and control technologies, you can implement:

- disengagement of the bridge associated with link Y (you will need to install a brake associated with link A; the bridge will be disengaged when this brake is turned on, while the row with the kinematic parameter k_1 will operate in the reduction gear mode);
- PDM blocking (you will need to install a blocking clutch C_0 for a row with kinematic parameter k_1; blocking is performed by simultaneously switching on C_1 and C_0);
- smooth measurement of the gear ratio between the output links (can be implemented using the principle of controlling the slipping of the disk friction elements T_1 and C_2 – this principle is described in more detail in [20, 21]).

The forbidden mode of operation when the engine is on is for such a mechanism the simultaneous activation of the controls T_1 and C_2. In this case, the Y link branch will be blocked and the axle associated with it will not be able to rotate, and the power flow from the engine will go to the X link branch, which, if there is a lack of grip weight, will cause the front axle wheels to slip. If the grip weight is sufficient, the machine will be able to move with catastrophic wear on the rear axle tires.

The kinematics of the mechanism is described by traditional equations obtained from the Willis formula [13-15]:

$$\begin{cases} \omega_A = k_1\omega_X + \left(1-k_1\right)\omega_0 \\ \omega_1 = k_2\omega_A + (1-k_2)\omega_Y \end{cases} . \tag{4}$$

Accept $\omega_0 = 1$.

To remove the kinematic uncertainty, we assume that the car moves without slipping: $\omega_X = \omega_Y$.

When the brake T_1 is turned on, the equality: $\omega_1 = 0$.

Then

$$\omega_X = \omega_Y = \frac{-\left(1-k_1\right)}{\left(k_1 + \dfrac{\left(1-k_2\right)}{k_2}\right)}\omega_0 = 1/2 . \tag{5}$$

Thus, when T_1 is turned on, kinematically the PDM performs the function of a symmetrical differential

In the case of inclusion of the blocking clutch C_2, the equality is fulfilled: $\omega_Y = \omega_A = \omega_1$. The device works like an asymmetrical differential.

The moment balance equations give the following systems describing the operation of planetary mechanisms:

$$\begin{cases} M_0 = 1 \\ M_X = -k_1 M'_A \\ M_0 = -\left(1-k_1\right)M'_A \end{cases} \tag{6}$$

$$\begin{cases} M'_A + M''_A = 0 \\ M_1 = -k_2 M''_A \\ M_Y = -\left(1-k_2\right)M''_A \end{cases} . \tag{7}$$

Turning on the brake T_1 is equivalent to the condition:

$$M_1 + M_{\text{T}1} = 0 . \tag{8}$$

The reaction moment acting on the crankcase when T_1 is turned on will be equal in absolute value and opposite in sign to the moment M_{T1}.

Then get:

$$M'_A = -M_0/(1-k_1) = 1/3\,; \tag{9}$$

$$M''_A = -M'_A = M_0/(1-k_1) = -1/3\,; \tag{10}$$

$$M_1 = -k_2\, M_0/(1-k_1) = 1/3\,; \tag{11}$$

$$M_{T1} = -M_1 = -1/3\,; \tag{12}$$

$$M_X = -k_1\left(-M_0/(1-k_1)\right) = -2/3\,; \tag{13}$$

$$M_Y = -(1-k_2)M_0/(1-k_1) = -2/3. \tag{14}$$

The distribution of moments over the output links is symmetrical. The mechanism is in balance.

When the blocking clutch C_2 is turned on, we get a picture of the distribution of moments characteristic of an asymmetric differential:

$$M_X = -k_1\left(-M_0/(1-k_1)\right) = -2/3\,; \tag{15}$$

$$M_Y = M_A = -M_0/(1-k_1) = -1/3. \tag{16}$$

If it is necessary to determine the values of the moments of resistance and the required traction forces on the drive wheels, traditional methods of the theory of wheeled and tracked vehicles are used [22-24].

For the design of the PDM, it seems possible to use the methods for choosing the number of teeth used in the development of planetary gearboxes [13, 25-27]. In the manufacture of parts and assemblies, it is possible to recommend technologies and materials that have been tested in the production of planetary gears for wheeled and tracked vehicles [13, 14, 25]. The issues of slip control for disc friction controls are described in a number of publications mentioned above [20, 21].

4 Practical implications and perspectives

The controlled interaxle power distribution mechanism, which makes it possible to realize the distribution of torque along the axes in ratios of 1:2 and 1:1, in addition to the case considered, can be used on trucks and transport tracked vehicles. At the same time, it seems possible to integrate such a mechanism into the transfer case of a mass-produced truck of the KAMAZ family.

5 Conclusions

The use of a controlled inter-axle power distribution mechanism will ensure a rational distribution of loads on the driving axles of a specialized forestry machine.

When developing a transmission, mass-produced units can be used, in particular, transfer boxes with an asymmetric differential used in the transmission of KAMAZ vehicles.

When designing and manufacturing new controlled power distribution mechanisms, it is advisable to use methods, materials and technologies that have been tested in the production of planetary gearboxes for tracked and wheeled vehicles.

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